AN ANALYSIS OF A RESIDENTIAL HEATING SYSTEM UTILIZING A SOLAR ASSISTED WATER-TO-AIR HEAT PUMP

John R. Dunbar

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ABSTRACT

The performance characteristics of a residential solar assisted water-to-air heat pump heating system are analyzed for State College, Pennsylvania. A realistic residence and solar assisted water-to-air heat pump system are modeled for this northern climate using the transient simulation computer code TRNSYS developed by the University of Wisconsin. The system is studied over a one month winter period, December, using actual hourly weather data for State College. The system is analyzed for both the cloudiest and clearest December weather recorded in the last 30 years. The collector area and storage tank capacity are varied and the effects on system performance are studied. Other parameters varied and studied are the collector/storage tank heat exchanger effectiveness and the room temperature.

The Pennsylvania State University

The Graduate School

Department of Mechanical Engineering

Analysis of a Residential Heating System Utilizing A Solar Assisted Water-to-Air Heat Pump

A Paper in

Mechanical Engineering

by

John R. Dunbar

Submitted in Partial Fulfillment
of the Requirements
for a Degree of

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NOMENCLATURE

Cmax - Maximum capacity rate of fluid (Btu/hr-F)

Cmin - Minimum capacity rate of fluid (Btu/hr-F)

COP - Coefficient of performance (dimensionless)

Cp - Specific heat of fluid (Btu/lb-F)

E - Heat exchanger effectiveness (dimensionless)

eg - Ethylene glycol-water solution

F′ - Collector geometry efficiency factor (dimensionless)

FR - Collector efficiency factor (dimensionless)

H - Solar radiation on a horizonal surface (Btu/hr-sqft)

HT - Incident solar radiation on a tilted surface (Btu/hr-sqft)

m - Mass flowrate of fluid (lb/hr)

NTU - Number of transfer units (dimensionless)

qu - Rate of useful energy collected by the collector (Btu/hr-sqft)

SCOP - System coefficient of performance (dimensionless)

Ta - Ambient air temperature (degrees F)

Tfi - Collector inlet fluid temperature (degrees F)

U - Overall thermal transmittance of storage tank (Btu/hr-sqft-F)

UA - Building thermal transmittance times exposed area (Btu/hr-F)

UL - Collector overall energy loss coefficient (Btu/hr-sqft-F)

w - Water

α - Absorptance of collector absorber plate (%)

ϵ - Emmissivity of collector absorber plate (%)

η - Solar collector efficiency (dimensionless)

ρ - Ground reflectance (dimensionless)

τ - Transmittance of collector cover plate (%)

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ABSTRACT

The performance characteristics of a residential solar assisted water-to-air heat pump heating system are analyzed for State College, Pennsylvania. A realistic residence and solar assisted water-to-air heat pump system are modeled for this northern climate using the transient simulation computer code TRNSYS developed by the University of Wisconsin. The system is studied over a one month winter period, December, using actual hourly weather data for State College. The system is analyzed for both the cloudiest and clearest December weather recorded in the last 30 years. The collector area and storage tank capacity are varied and the effects on system performance are studied. Other parameters varied and studied are the collector/ storage tank heat exchanger effectiveness and the room temperature.

CHAPTER 1

1.1 Introduction

With the energy crisis worsening and conventional heating fuels becoming scarce and expensive a need exists for the development of alternative heating fuels and systems. Solar energy which is "free" and readily available is a promising source of building heat. Unfortunately for northern climates direct solar heating of buildings can provide only a small portion of the total heating requirements without extremely large collector areas and heat storage capacity[1].* The solar energy can however be used to augment conventional heating systems or used to improve the performance of the system, as with a heat pump. The heating capacity of the heat pump decreases as the evaporator temperature (i.e. the outdoor ambient air temperature) decreases. At the same time the heating load of the residence increases. The point at which the heat pump capacity equals the heating load is called the balance point. The heat pump is an economical heat source as long as it can operate at a temperature above its balance point. In cold northern climates the evaporator inlet temperature of the standard air-to-air heat pump installation is often below the balance point. For water-to-air heat pumps an

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* Numbers in brackets refer to references listed at the end of this paper.

existing warm water source (temperature above the balance point) for the heat pump evaporator is difficult to find in northern climates particularly for residential installations. Solar energy can be used easily to provide this warm temperature source for the heat pump evaporator since only a low grade temperature (50-100 degrees F) is required.

Over the past several years many papers and studies have been done on the evaluation of solar energy heat pump systems. The earliest research on solar assisted heat pumps was begun by the American Gas and Electric Service Corporation in 1949 in a 5 year study which used a high temperature storage system with the heat pump coupled directly to the collector[2]. In 1952 R. C. Jordan and S. L. Threlkeld published a paper in which the potential of this system on the basis of some experiments was described[3]. Since that time the solar assisted heat pump system has been improved upon and extensively studied. The effects of geographical location and variations in the system design or component design have been addressed in the present literature[4-6].

The study described in this paper will analyze the performance of a solar assisted water-to-air heat pump residential heating system in State College, Pennsylvania (latitude 40.5 degrees N). A realistic residence and solar

assisted water-to-air heat pump system are modeled to study the performance characteristics of such a system in a northern climate. The transient simulation computer code TRNSYS developed by the University of Wisconsin and as modified at the Pennsylvania State University is used. The system is studied over a one month winter period, December, using actual hourly weather data for State College. The system is analyzed for both the cloudiest and clearest December weather recorded in the last 30 years. The collector area and storage tank capacity are varied and the effects on system performance are studied. Other parameters varied and studied are the collector/storage tank heat exchanger effectiveness and the room temperature.

CHAPTER 2

2.1 Description of Residential Structure

For investigation and modeling purposes the solar assisted heat pump system was assumed to be installed in a single family residence. It is a two story structure with a full basement and attached two car garage. The living area is approximately 2200 square feet with 8 foot ceilings throughout. The solar collectors are located on the south slope of the roof and are tilted at 50.5 degrees (latitude + 10 degrees), the recomended [2] optimum angle for heating applications. A floor plan of the residence is shown in Figure 1. The residence is located in State College, Pennsylvania, a small university oriented city situated in the center of the state.

An energy conserving design has been utilized to minimize the required heating load. The house walls are built of wood frame construction with four inch face brick. All windows are double pane glass and a storm door is installed on the front door. Three and one half inches of R-11 insulation is installed in the walls and six inches of R-22 insulation is placed in the second floor ceiling. A more complete list of the construction details can be found in Appendix A.

TWO-STORY RESIDENCE

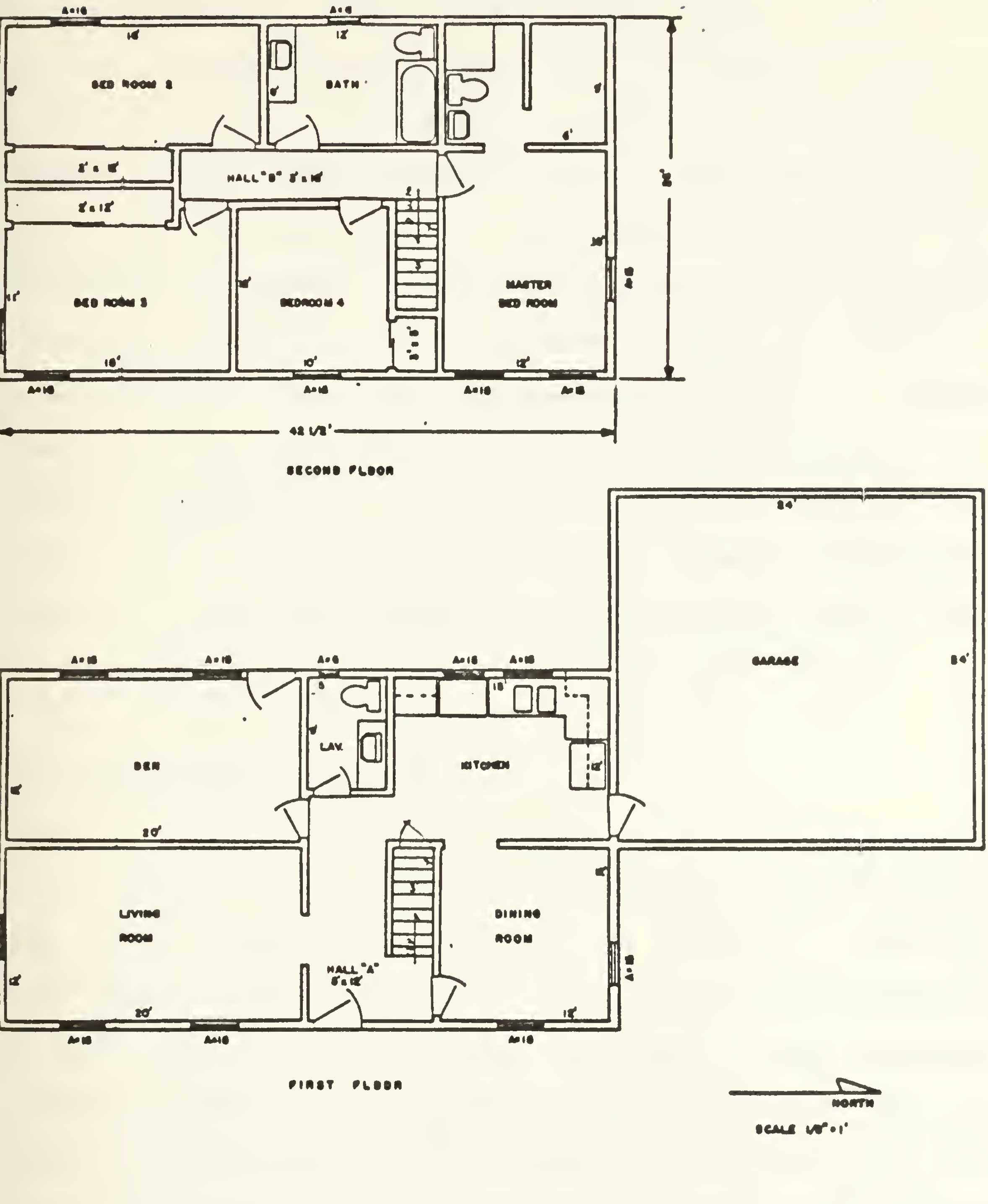

FIGURE 1 - Floor plan of residence utilizing a solar assisted heat pump system in State College, Pa.

In order to size and select the appropriate heat pump it was necessary to calculate a design summer cooling load as well as the winter heating load. The heating load was calculated using the general procedures set forth by ASHRAE[3] using ASHRAE tabulated data. The resulting design heat loss was approximately 32222 Btu/hr. From this the building UA (thermal transmittance X exposed area) was found to be 511 Btu/hr-F which indicates a very well insulated house. The cooling load was determined using the ASHRAE simplified procedures for residential calculations[4] and ASHRAE tabulated data. The cooling load requirement was found to be approximately 1.4 tons. Summary calculation sheets for both the heating and cooling loads can be found in Appendix A.

2.2 System Description

A schematic diagram of the residential solar assisted heat pump heating system is shown in Figure 2. The major components of the system are a flat plate solar collector, storage tank and a water-to-air heat pump. Other equipment includes pumps, heat exchanger, a relief valve and auxiliary electric heating which is an integral part of the heat pump. The major components are discussed in more detail in the following paragraphs.

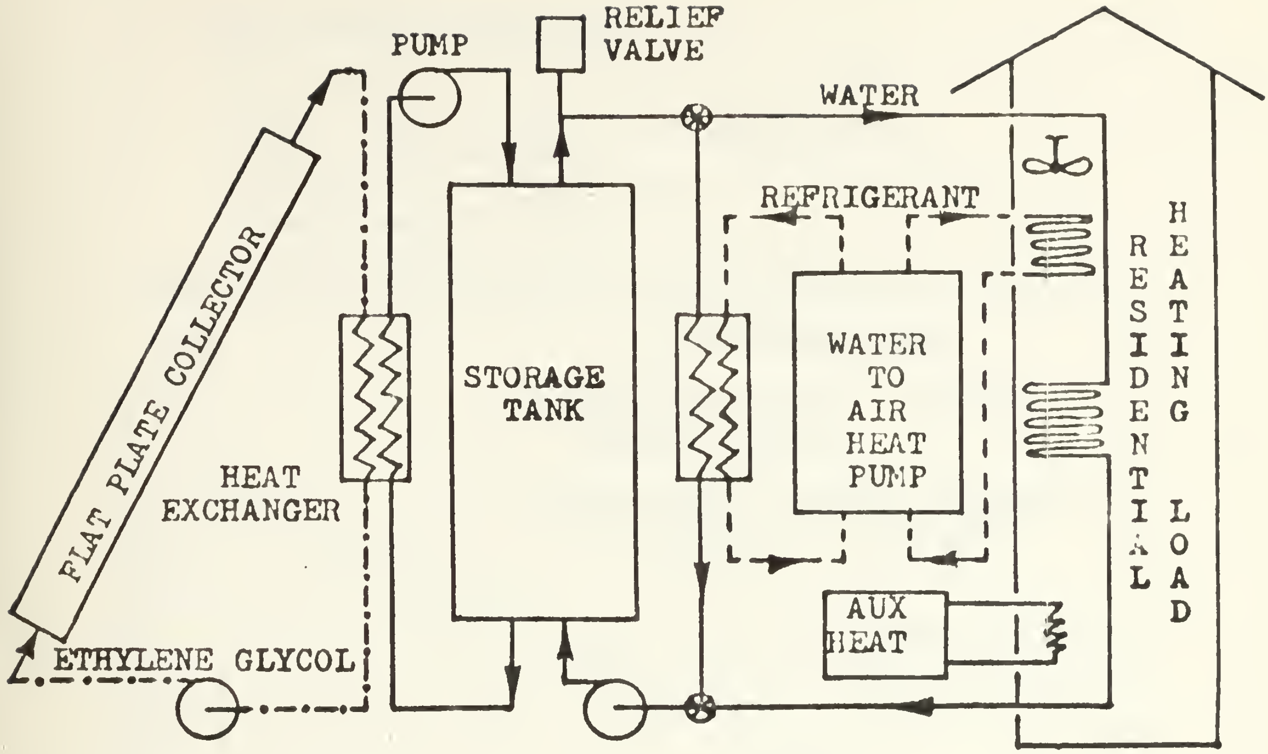

FIGURE 2 - Schematic of solar assisted heat pump heating system

A. <u>Flat Plate Collector</u> - The flat plate solar collector chosen for the simulation was a Pittsburg Plate Glass(PPG) Model CD414. The PPG collector features weather sealed construction with two tempered glass cover plates and an all copper absorber plate with a non-selective coating. The collector is especially designed for applications where the collector area is limited or where large temperature differentials exist between the ambient air and the absorber plate. Each collector panel is approximately 20 square feet. The collector array uses a 40% ethylene glycol-water solution as the circulated heat transfer medium. This provides protection from freezing to -12 degrees F.

ADDITIONAL PARAMETERS

Wind Velocity: 0-5 mph
Fluid: 40% ethylene glycol solution
Cp = .792 Btu/lb-F
Flowrate = .025 gpm/sqft
Tube Diameter = .5 in
Tube Spacing = 3 in

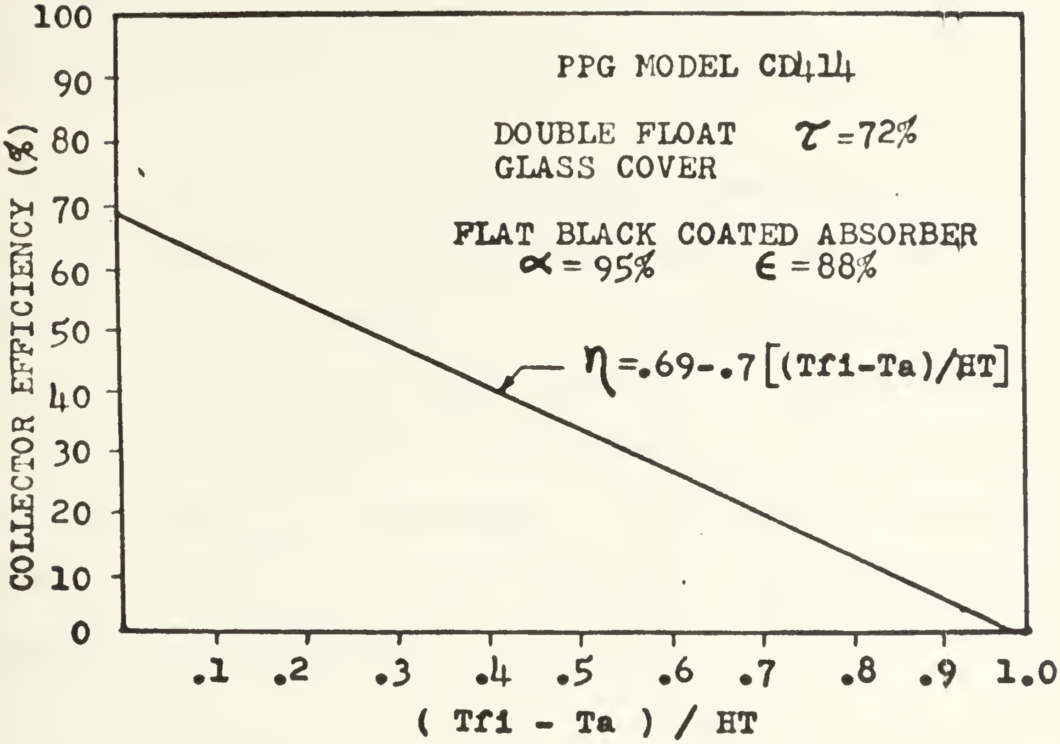

FIGURE 3 - Efficiency of the PPG flat plate solar collector

The performance of this flat plate collector can be described by a number of parameters. Perhaps the most basic performance parameter is the collector efficiency( ) which for the PPG collector is

$$=.69-.7[(Tfi-Ta)/HT] \qquad (1)$$

where Tfi=inlet fluid temperature(degrees F)
Ta =ambient air temperature(degrees F)
HT =incident radiation per unit area on the collector surface(Btu/hr-sqft)

The efficiency equation is shown graphically in Figure 3 along with a number of other parameters describing the construction of the PPG flat plate collector. These parameters include the transmittance (τ) of the cover, the absorptance(α) of the absorber plate and the emmissivity(ϵ) of the absorber plate.

Three other useful parameters describing the PPG collector are the Collector Efficiency Factor(FR), the Collector Overall Energy Loss Coefficient(UL) and the Collector Geometry Efficiency Factor(F′). These parameters are related to the Rate of Useful Energy Collected per Unit Area of Collector(qu) according to the following expression developed by Hottel and Woertz[5] and extended by Whillier[6] and Bliss[7].

$$qu=FR[HT(\tau\alpha)-UL(Tfi-Ta)] \qquad (2)$$

Since η =qu/HT equation(2) can be rewritten as

$$=FR(\tau\alpha)-[FR*UL(Tfi-Ta)/HT] \qquad (3)$$

Comparing this to the collector efficiency equation for the PPG collector, it follows that

$$FR(\tau\alpha)=.69 \quad \text{and} \quad FR*UL=.7 \qquad (4)$$

Substituting known values for the transmittance and absorptance

$$FR=.69/(.72)(.95)=1.0 \qquad (5)$$

thus

$$UL=.7\ Btu/hr\text{-}sqft\text{-}F \qquad (6)$$

Finally, using UL, the Collector Geometry Factor(F′) can be obtained from graphs developed by Duffie and Beckman[8]. The value of F′ for the PPG collector is .99.

B. Storage Tank - An insulated steel underground tank containing water is used for thermal storage. The overall thermal transmittance(U) of the tank is .05 Btu/hr-sqft-F. The tank is cylindrical in shape with a constant length to diameter ratio of three(as the volume is varied in the simulation). A tube-in-shell heat exchanger with an assumed effectiveness of 70% is used to transfer heat from the collector loop to the storage tank.

C. Water-to-Air Heat Pump - An American Air Filter(AAF) water-to-air heat pump Model HW19 (1.5 tons) was selected for the residence. As is standard practice the heat pump was selected to meet the cooling load requirements of the building. It is recognized that in harsh weather the heat pump will be unable to provide 100% of the heating load therefore the AAF heat pump is equipped with a parallel

auxiliary electric resistance heater to provide the deficit. The AAF Model HW19 water-to-air heat pump is rated to provide 600 CFM of heated air to the house. The water flow on the water side of the heat pump is 5 gpm. Table 1 is the heating performance characteristics of the Model HW19 for maintaining the room temperature at 70 degrees F.

TABLE 1 - Heating performance characteristics of AAF Model HW19 heat pump

EVAP INLET TEMP (degrees F)	HEAT ABSORBED (MBtu/hr)	HEAT REJECTED (MBtu/hr)	POWER INPUT (kw)	POWER INPUT (MBtu/hr)	COP
40	11.5	18.0	1.80	6.14	2.93
50	13.3	20.7	2.06	7.03	2.94
60	15.2	23.2	2.30	7.85	2.96
70	17.1	25.9	2.53	8.63	3.00
80	19.0	28.4	2.76	9.42	3.01
90	20.9	31.1	2.99	10.20	3.05
100	22.8	33.7	3.22	10.99	3.07
110	24.7	36.3	3.45	11.77	3.08

2.3 Operation of System

The solar assisted heat pump system is designed to supply the complete heating requirements of the residence with minimal energy consumption. The required heat is provided by one of three modes. These modes are direct heating from storage, heat pump heating and auxiliary electric resistance heating.

The collector loop uses a 40% ethylene glycol-water mixture to prevent freezing during low ambient temperature conditions. The mixture is pumped through the collector/heat exchanger loop at a rate of .025 gpm/sqft of collector. The pump is controlled by a control mechanism which turns the pump off when the temperature of the fluid leaving the collector is less than the storage tank temperature. The pump resumes operation when the temperature of the fluid in the collector reaches the storage tank temperature plus 10 degrees F. This control sequence prevents the loss of thermal energy from the tank by reradiation through the collector at night or cloudy days when solar insolation decreases to a minimum.

When the house thermostat calls for heat and the storage tank temperature is greater than or equal to 110 degrees F water from the tank is pumped through a heating coil in the house air duct while the house air fan circulates warm air through the house. This is direct heating from the storage tank. If heating is required and the storage tank temperature is below 110 degrees F the direct heating mode is disabled and the control valves circulate the water through the heat pump heat exchanger. The heat pump now provides heat to the house. On occasion the capacity of the heat pump will be unable to meet the house heating requirement in which case the built-in house air stream electric resistance heater of the heat pump will

provide the additional required heat. If the storage tank temperature falls below 40 degrees F and heating is still required then the heat pump is shut down and all the required heating is provided by the electric resistance heater.

If the ambient air temperature exceeds 65 degrees F the heating requirements decrease to zero and all heating modes become inoperative.

2.4 Computer Model

The solar assisted heat pump system shown in Figure 2 was modeled using the Transient Simulation Program (TRNSYS) developed at the University of Wisconsin. The program allows one to make a transient (time dependent) simulation of the overall system by interconnection of individual component models. The TRNSYS modeling components used are shown in Figure 4 which is a flow diagram of the computer model.

The reader component is used to input hourly weather data. The simulation is run for one month of the heating season. Only ambient temperature and solar insolation (measured on a horizontal surface) data are needed for the simulation. Actual weather data obtained from the Pennsylvania State University weather station for the months

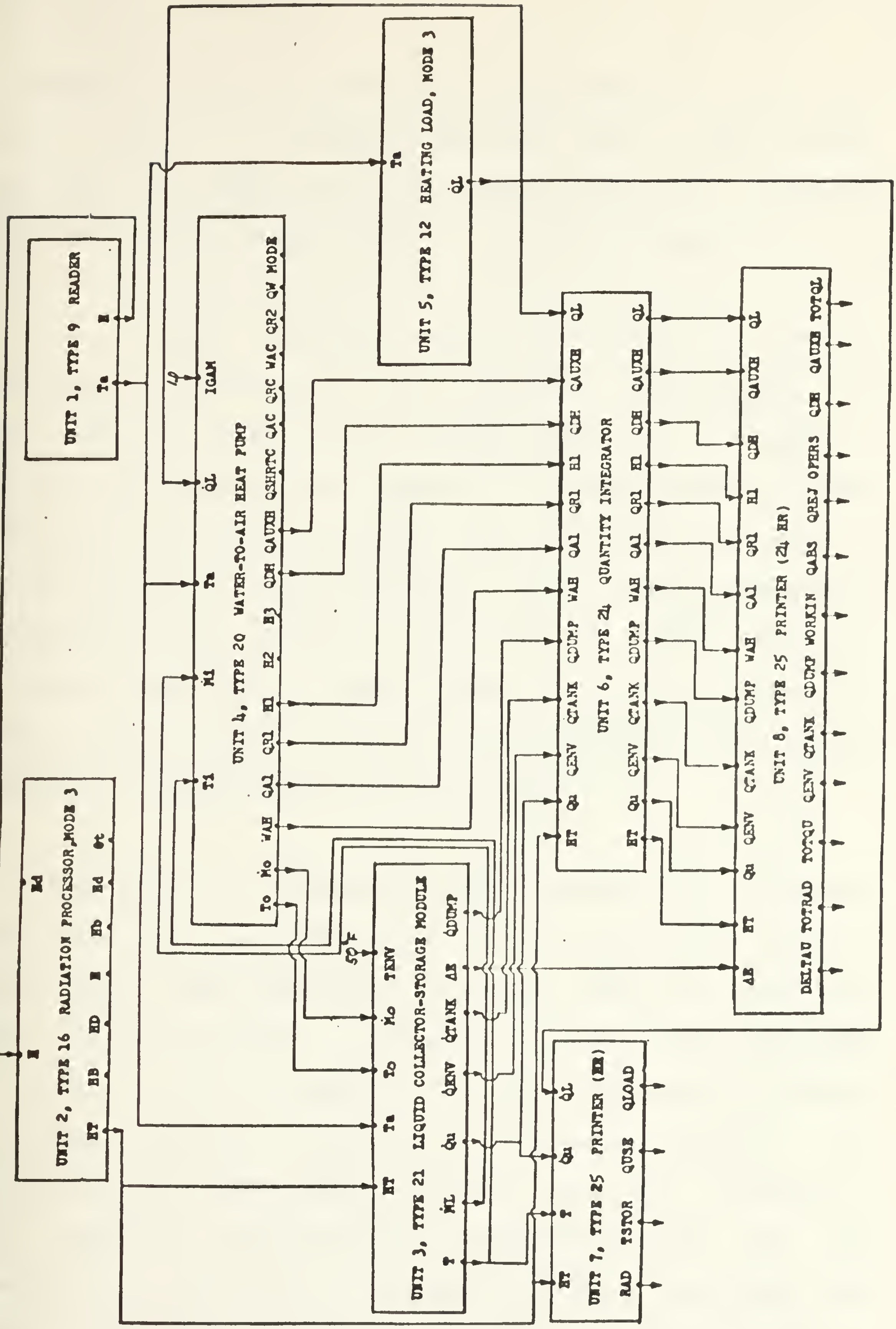

FIGURE 4 - TRNSYS computer model flow diagram (Terms are defined in Appendix B)

of December 1960 and December 1972 are used. For the past thirty years of available weather data, these years represent the highest and the lowest values for December solar insolation, respectively, for State College,Pa.

The radiation processor component converts the solar insolation or radiation on a horizontal surface (H), to solar radiation on a tilted surface (HT). The collector is tilted at an angle of 50.5 degrees. By using Mode 3 of the radiation processor component the total radiation on a tilted surface is a combination of the beam radiation, diffuse radiation and reflected radiation. The radiation processor calculates these values from the horizontal surface radiation, the collector orientation and the ground reflectance which is .2 for bare ground.

The liquid collector-storage module is a single component used to model the solar collector, ethylene glycol/water heat exchanger, storage tank and associated pumps and relief valve. A schematic of the module is shown in Figure 5. The combining of these components into one subsystem reduces computation time and expense. The collector fluid, 40% ethylene glycol solution, is pumped at the collector manufacturers recommended rate of .025 gpm/sqft of collector. Across the heat exchanger the capacity rate ratio is held at 1.0 [i.e. $(mCp)_{eg} = (mCp)_w$].

The initial storage tank temperature at the beginning of December is assumed to be 70 degrees F. It turns out that any initial temperature in this range gives nearly identical results over the one month period. The relief valve is set to maintain the tank temperature below 212 degrees F.

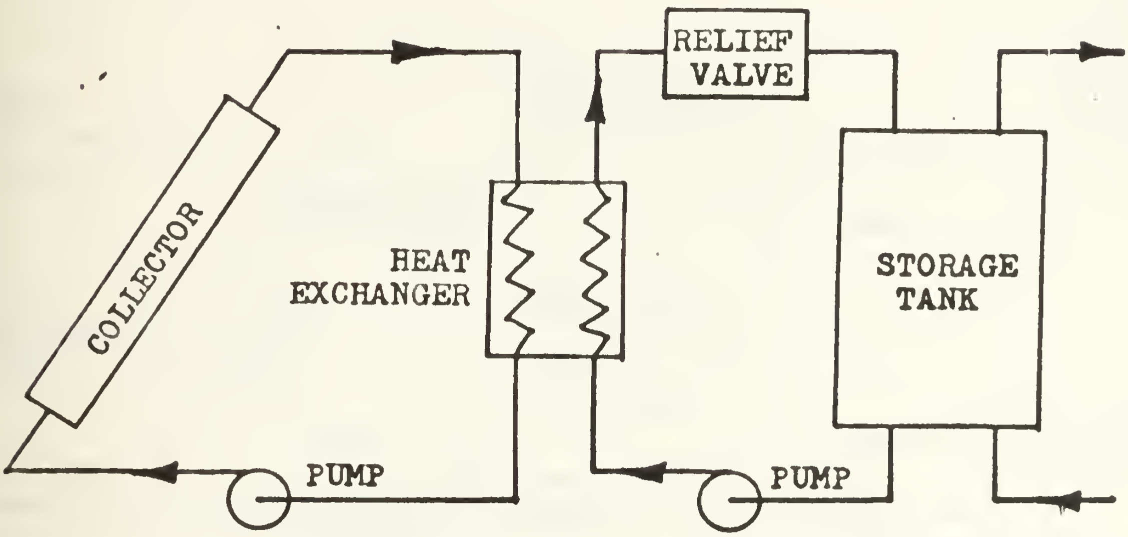

FIGURE 5 - TRNSYS liquid collector/storage module

The water-to-air heat pump component uses actual heat pump performance characteristics. The model is quasi-steady state in nature. That is, at any instant in the "transient" simulation, the heat pump performs as if it were at steady state, having for the moment constant source and sink temperatures and hence a unique heat addition, heat rejection and work input. The heat pump is modeled to operate in the heating mode only. A parallel auxiliary

electric heating element is included to add just enough energy to meet the heating load when the heat pump capacity is less than the load or when the storage tank temperature falls below 40 degrees F. Figure 6 is a schematic of the water-to-air heat pump component.

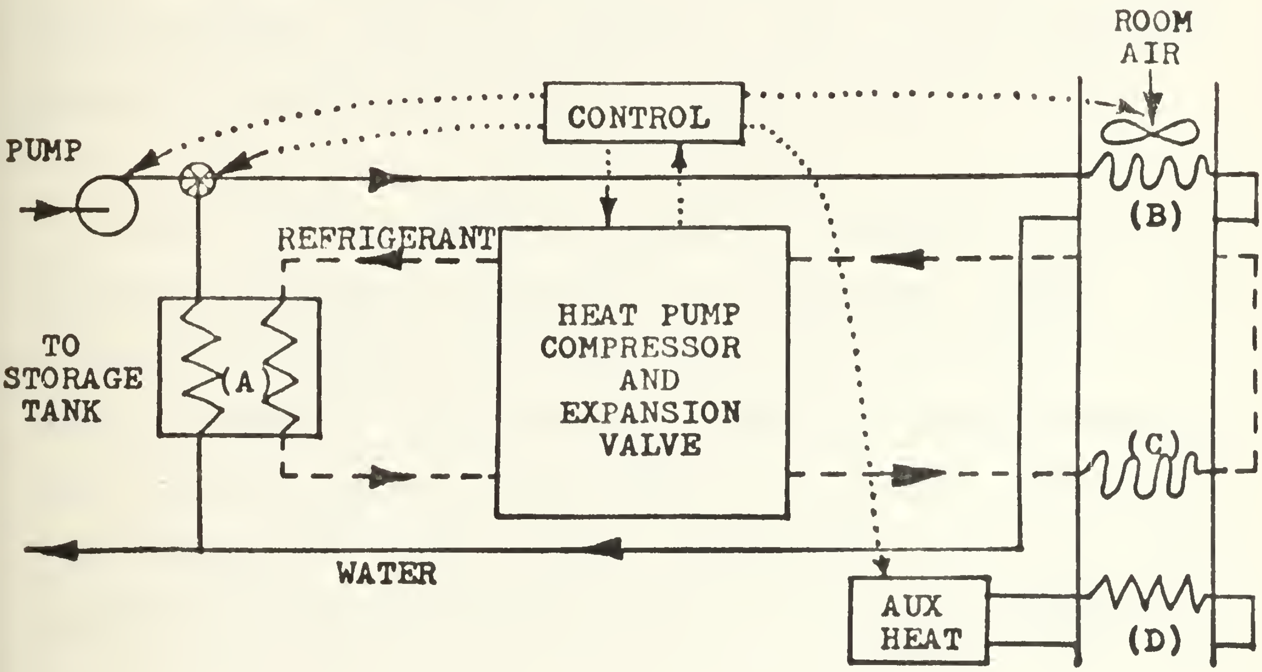

FIGURE 6 - TRNSYS water-to-air heat pump component

The heating load component determines the heating load of the residence by use of the energy/degree day concept. The concept is extended to estimate the hour-by-hour heating load by multiplying the heating requirement of the residence (energy/degree-hour) by the difference between the required room temperature and the outdoor ambient temperature.

The quantity integrator simply keeps a continuous total of various system outputs throughout the simulation. It works similiar to a KWH meter which continuously totals the amount of electrical energy consumed.

The printer components control the display of the system outputs. In this simulation some of the outputs are printed at hourly intervals and some at daily intervals.

Each of the TRNSYS components requires a set of defining parameters and inputs to produce an output. The parameters describe the operating characteristics of the system component being modeled. Many of these parameters have been discussed previously. A complete list and description of the typical parameters used in this simulation for each component can be found in Appendix B.

CHAPTER 3

3.1 Presentation and Discussion of Results

The collector efficiency, which is a measure of the useful solar energy being transferred to the heating system, is shown in Figures 7 and 8 for various values of collector area and storage capacity. The collector efficiency is maximum where the collector area is small and the storage capacity is large. This can be shown also by the collector efficiency equation when one realizes that the inlet fluid temperature to the collector (from storage) will be a minimum when the ratio of collector area to storage capacity is a minimum. In other words looking at equation (1) again,

$$\eta = .69-.7[(Tfi-Ta)/HT], \qquad (1)$$

low values of Tfi (resulting from a small collector area to storage capacity ratio) will reduce the second term on the right hand side resulting in a larger value for the collector efficiency. The collector efficiency increases for increases in the storage capacity and decreases in the collector area since both of these trends decrease the temperature of the fluid entering the collector. Figure 7 also shows that increasing the storage capacity beyond about 2 gal/sqft of collector area is of little benefit since the collector efficiency remains relatively constant.

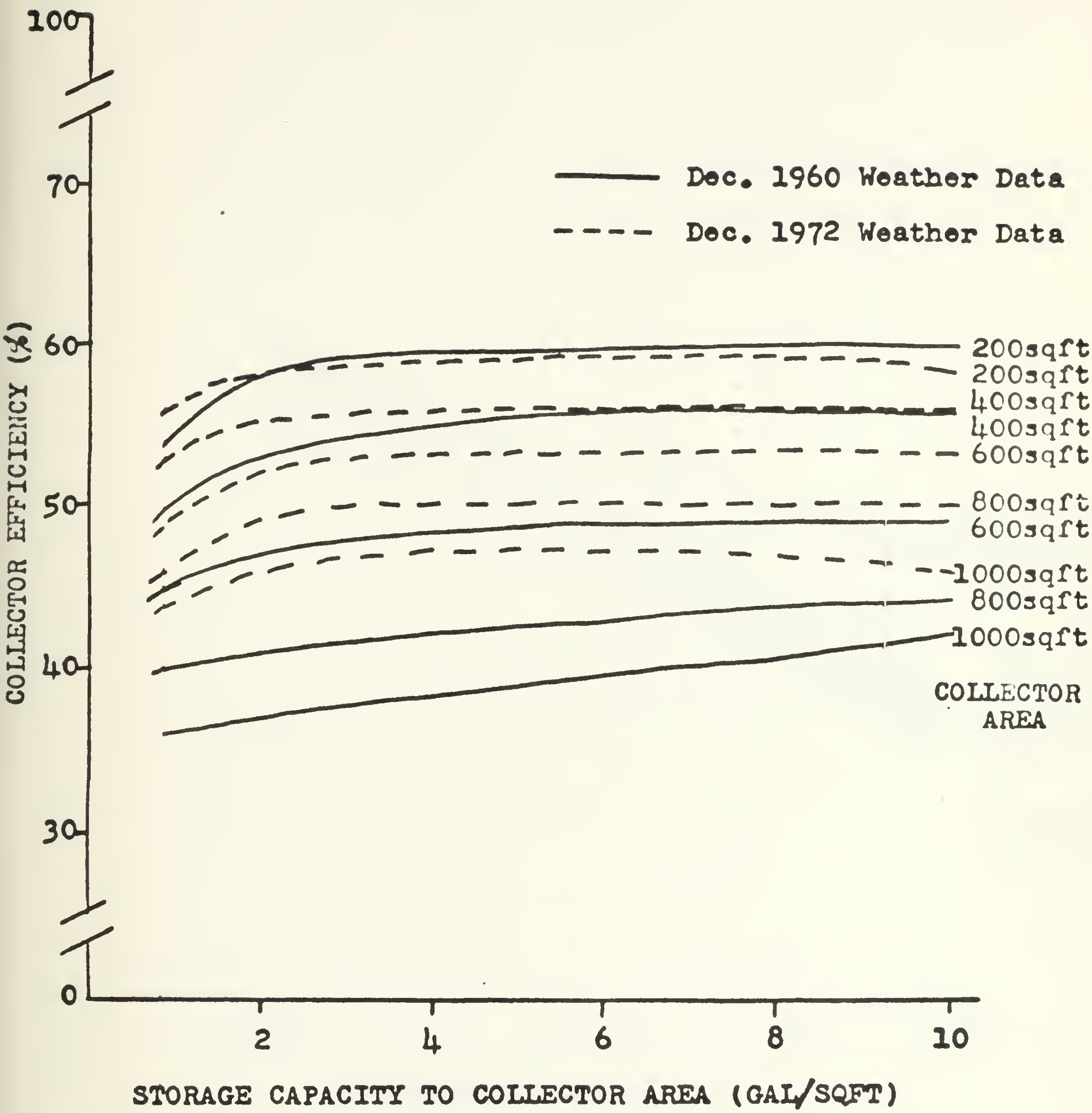

FIGURE 7 - Collector efficiency vs storage capacity to collector area for solar assisted heat pump system in State College, Pa.

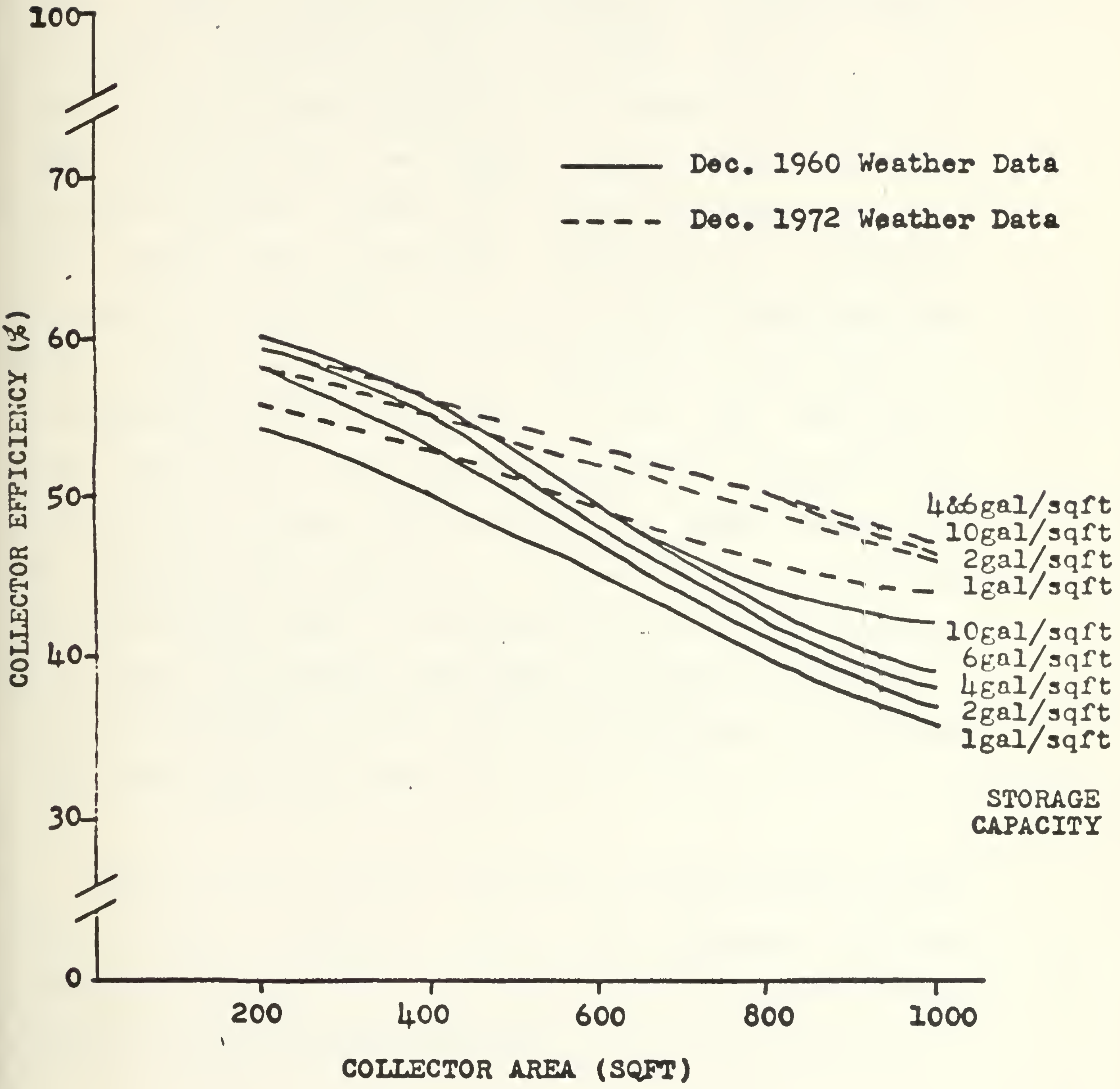

FIGURE 8 - Collector efficiency vs collector area for solar assisted heat pump system in State College, Pa.

The effect of weather on the collector efficiency can be seen by comparing the curves for 1960 and 1972 in Figures 7 and 8. The 1972, or minimum solar insolation, curves reflect higher collector efficiencies for identical system parameters (collector area and storage capacity). This result is not as easy to predict from equation (1) since all three variables on the right side change. From the graphical results, however, it can be concluded that for poor weather the predominant effect on the efficiency equation is a decrease of the second term on the right hand side. This implies that the magnitude of the drop in entering fluid temperature must be larger than the drop in solar insolation, indicating relatively low entering fluid temperatures (storage tank temperatures) would be expected for poor weather conditions. Figure 7 also shows that weather extremes have a negligible effect on the collector efficiency of systems with small collector area and a significant effect on the collector efficiency of systems with large collector area.

Values of the System Coefficient of Performance (SCOP) for various collector area and storage capacity configurations are shown in Figures 9 and 10. The SCOP is defined by the following equation:

$$SCOP = \frac{\text{total heating load}}{\text{electric energy used by heat pump \& aux heater}} \qquad (7)$$

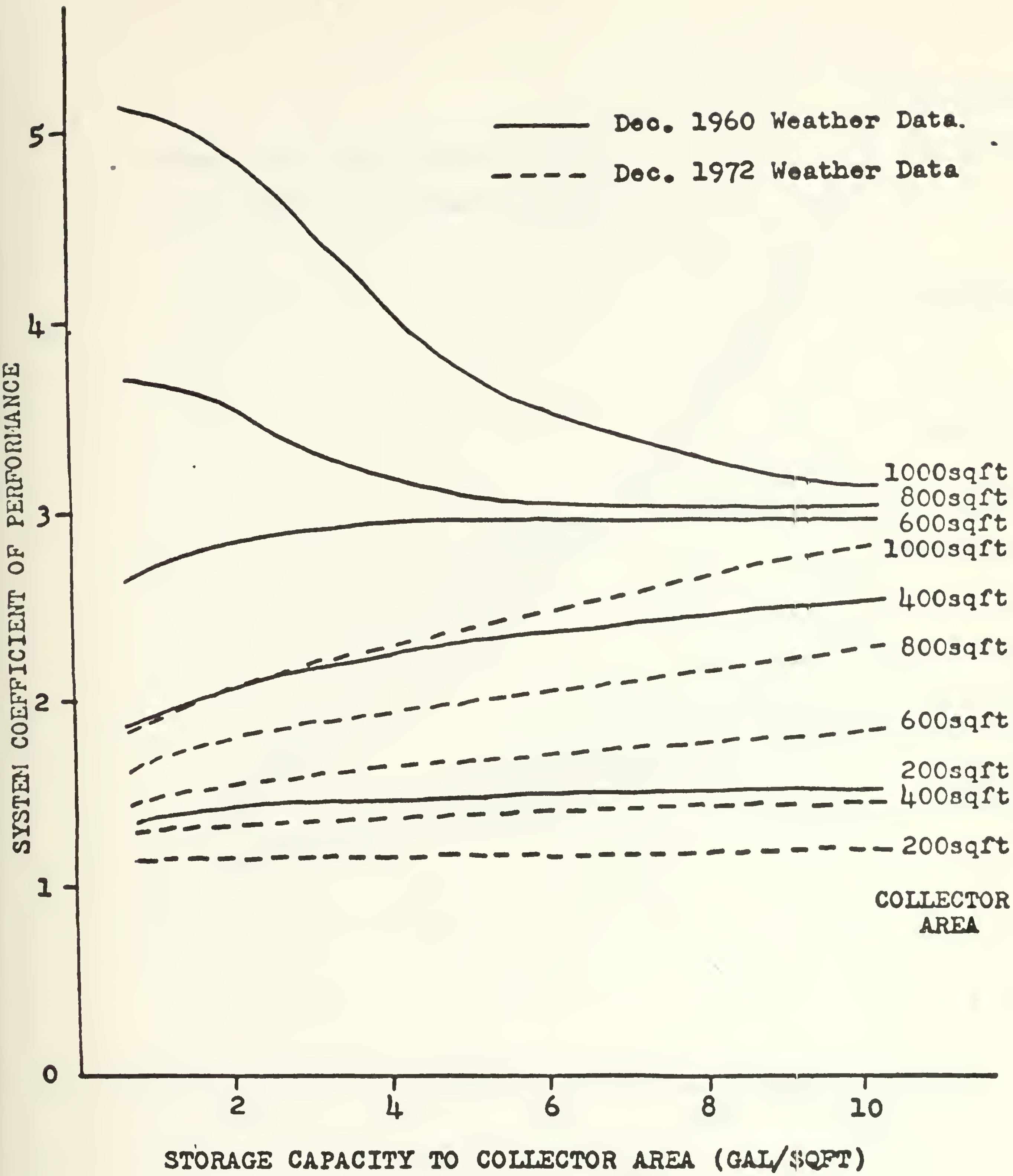

FIGURE 9 - System coefficient of performance vs storage capacity to collector area for solar assisted heat pump system in State College, Pa.

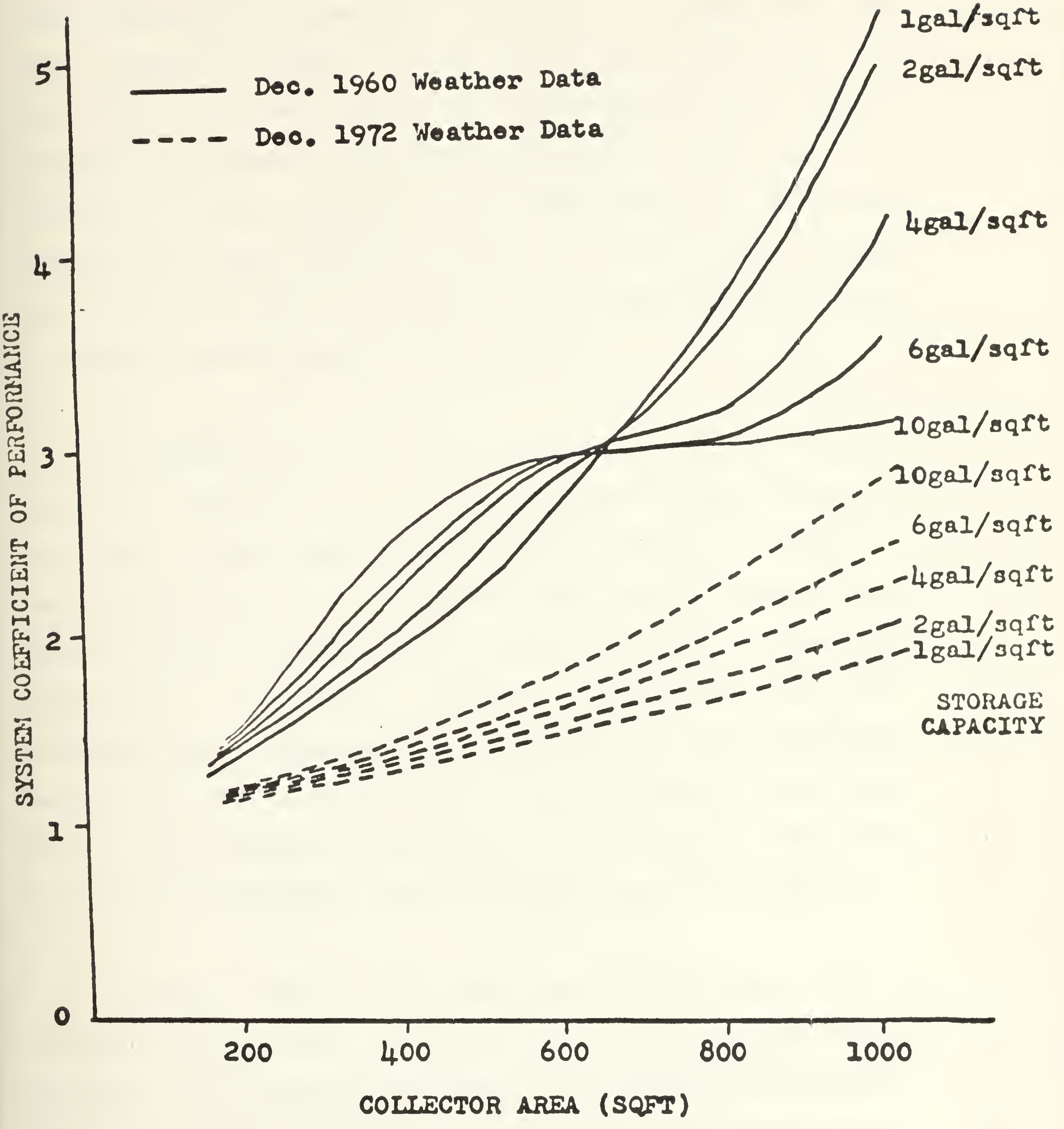

FIGURE 10 - System coefficient of performance vs collector area for solar assisted heat pump system in State College, Pa.

Following conventional practice, the energy required for fans and pumps is considered negligible in comparison with other energy inputs. Since the total heating load requirement for the residence is independent of the heating system configuration, the SCOP will depend only upon the amount of energy used to run the heat pump and the energy required for the auxiliary heater. The required energy input from these two sources is reduced, however, when there is a contribution from direct heating.

In Figures 9 and 10 it can be seen that all the curves approach a SCOP of 3.0 as the overall system size (collector area and storage capacity) increases. This is because as the storage capacity to collector area ratio increases the system becomes more and more dependent upon the heat pump alone (i.e. the direct heating contribution and the auxiliary heat contribution to the heating load decrease). Specifically, the curves converge to a value of 3.0 since that is the average heating COP of the AAF heat pump over its range of evaporator inlet (storage tank) temperatures.

Figures 9 and 10 both show that the maximum SCOP is obtained with a large collector area and a small storage capacity. The factor responsible for driving the SCOP past 3.0 (the average heating COP of the heat pump) is the direct heating contribution. Without a direct heating contribution

the SCOP will be limited to a maximum value equal to the heating COP of the heat pump. Also from Figure 10 it is seen that for a SCOP greater than 3.0 the solar assisted heat pump system must have as a minimum a collector area greater than about 650 sqft.

The collector area of 650 sqft in Figure 10 is also important in that it is the approximate collector area for which all the different storage capacity curves resulting from 1960 weather data intersect each other at a SCOP equal to 3.0. The significance of this intersection is that it represents the point at which the total residence heat requirement is provided by the heat pump and system performance is independent of the storage capacity. This means that at 650 sqft of collector area the energy input from the solar collector array is equal to the energy withdrawn from the storage tank by the heat pump, resulting in neither the benefit of a direct heating contribution or the requirement for auxiliary heat. By extrapolation it is estimated that a similiar intersection point will occur for the storage capacity curves resulting from 1972 weather data at about 1800 sqft. Since 1960 and 1972 represent the extremes of solar insolation it is logical to expect that for the average year the intersection point will be somewhere between 650 and 1800 sqft, say 1200 sqft. Thus, based on a year of average weather the solar assisted heat

pump system for State College would require a collector area of 1200 sqft or greater to obtain a SCOP greater than 3.0 (the heat pump heating COP).

For collector areas below the intersection point the residence heat load is supplied essentially by the heat pump and auxiliary heat and the SCOP increases as the storage capacity increases. For collector areas above the intersection point the residence heat load is supplied by the heat pump, auxiliary heat and direct heat and the SCOP decreases as the storage capacity increases.

This situation suggests that a two tank storage system would be beneficial. A smaller high temperature tank would be charged first by the solar collectors and be used for direct heating. A second larger tank in the medium temperature range would be charged with the excess solar energy and be used as a heat source for the heat pump. This combination would allow the direct heating contribution to be maximized. Whether or not the advantages of a two tank system outweigh the added cost and complexity would have to be studied in more detail. Figures 9 and 10 also indicate that for a single storage tank system the SCOP is more strongly influenced by the collector area than by the storage capacity, since in general the SCOP increases more for an increase in collector area than for an increase in storage capacity.

Comparing the curves resulting from 1960 weather data (good) and 1972 weather data (poor) in Figures 9 and 10 shows that the weather data used for a solar assisted heat pump system simulation has a major effect on the resulting system performance characteristics. The SCOP obtained for a given collector area-storage capacity configuration can vary by more than 250%. If non-representative weather data are used in the design calculations, a system design could result which is inadequate for one situation or overdesigned for another.

Figures 11, 12 and 13 show the precentage contribution made by the heat pump, direct heat and auxiliary heat to the total heating load for a range of collector areas and storage capacities. For a given collector area the heat pump contribution increases as the storage capacity increases. Accordingly, the percentage of the total heat load supplied by auxiliary heat decreases. Looking at the bar graphs for 1960 and 1972 (representing weather extremes) it is seen that poor weather, as intuitively expected, increases the auxiliary heat requirements. The amount of increase is most severe around collector areas of 600 sqft where it averages about 40%. The percentage contribution made by the heat pump, for a fixed storage capacity to collector area ratio, peaks in the middle range of collector areas. As predicted previously by Figures 9 and 10, the

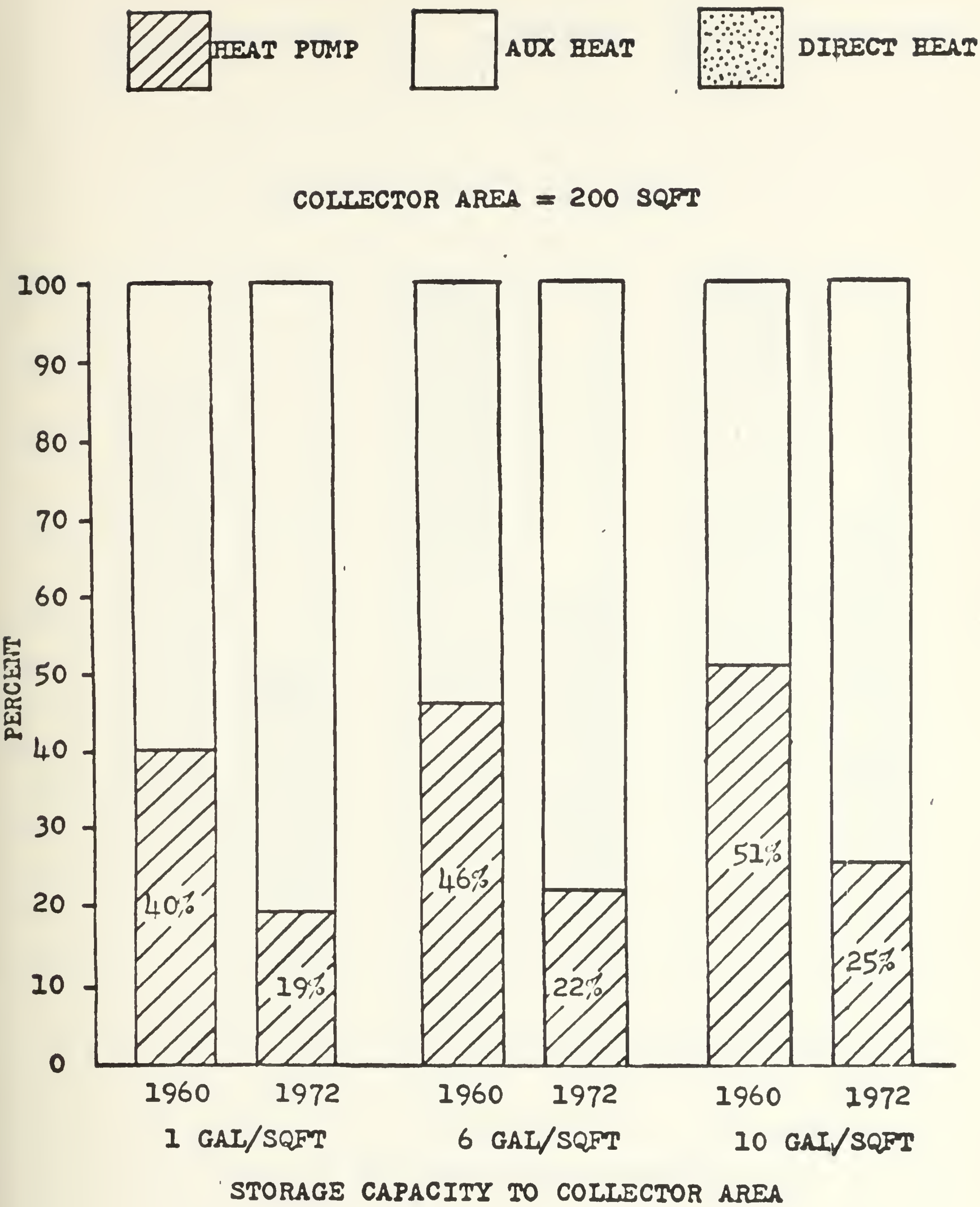

FIGURE 11 - Source of heat for solar assisted heat pump system in State College, Pa. at 200 sqft of collector area

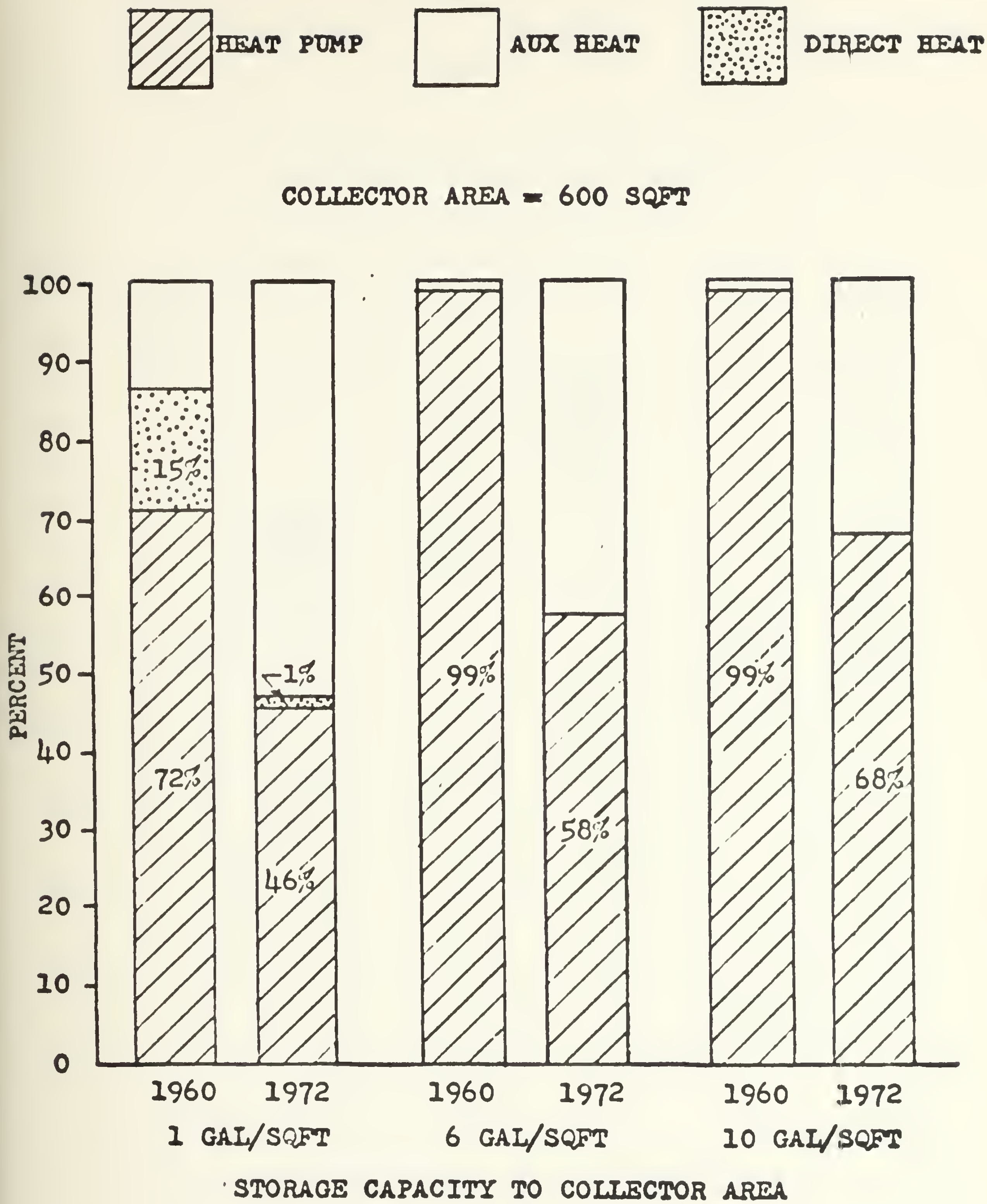

FIGURE 12 - Source of heat for solar assisted heat pump system in State College, Pa. at 600 sqft of collector area

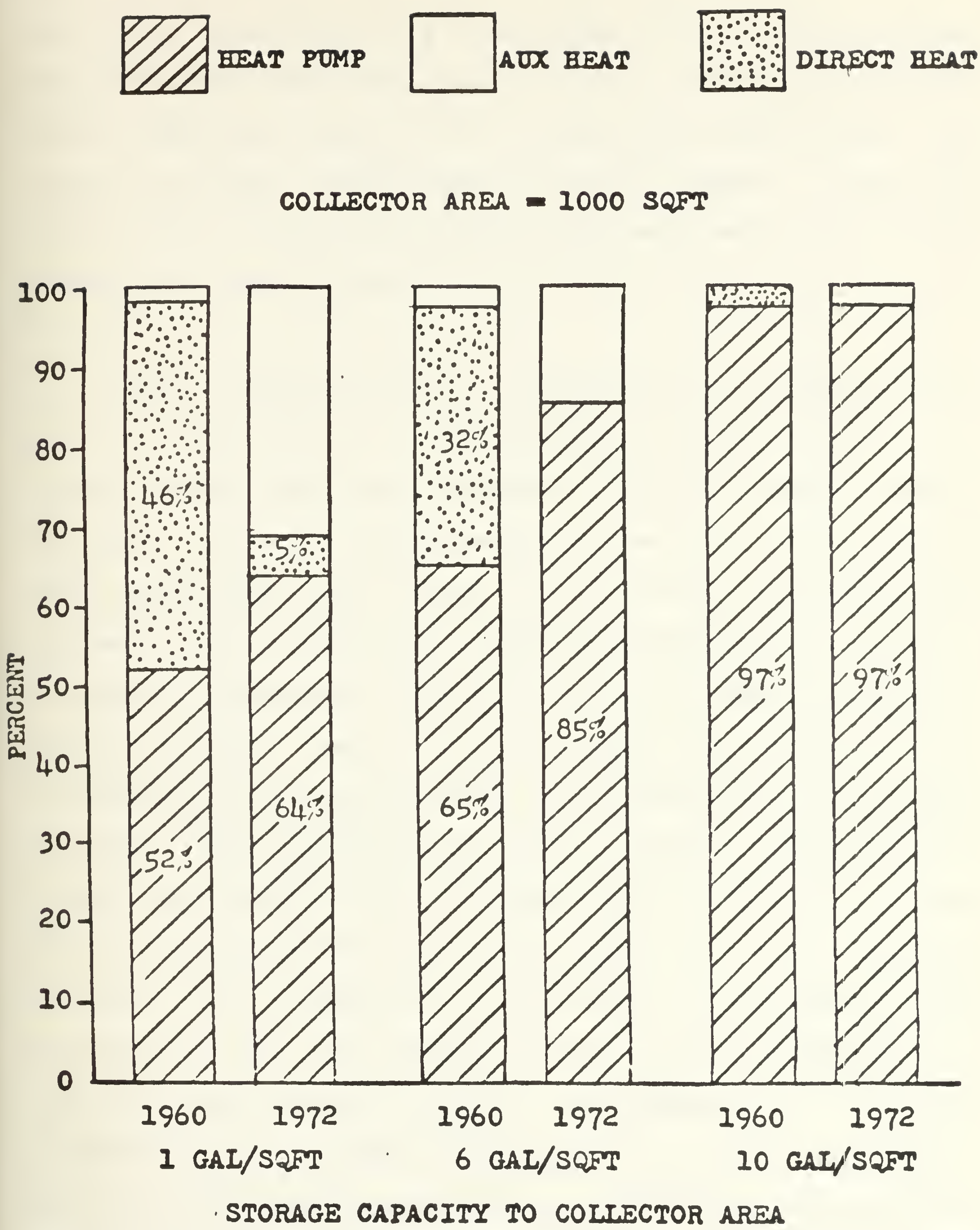

FIGURE 13 - Source of heat for solar assisted heat pump system in State College, Pa. at 1000 sqft of collector area

heat pump contribution is near 100% for a collector area near 650 sqft (for 1960 weather data). At large collector areas (1000 sqft, Figure 13) the heat pump contribution is smaller for good weather than for poor weather because the direct heat contribution lessens the heat requirement to be provided by the heat pump.

Direct heating becomes a significant contribution when the collector area is large and the storage capacity is small because the large amount of collected solar energy only has to heat a small amount of water, resulting in a high temperature level in the storage tank. Decreasing the collector area causes the percentage contribution of direct heating to decrease rapidly so that for small collector areas the direct heating contribution is non-existent. For a sufficiently large collector area, increasing the storage capacity will decrease the direct heating contribution. Also, poor weather will decrease the likelihood of any direct heating contribution. In Figure 13, for a storage capacity of 1 gal/sqft of collector, the direct heat contribution for poor weather (1972) is only 5% compared to 46% for good weather (1960). As indicated by previous figures, the bar graphs of Figures 11, 12 and 13 show that the best performimg configuration of the solar assisted heat pump system is a large collector area with a small storage capacity, mainly because of the advantage of more direct heating.

Table 2 lists the energy usage of a system with 1000 sqft of collector area and a storage capacity of 1 gal/sqft of collector. Energy data is given for both "good" and "poor" weather. It is clear from the percent deviations listed that weather has a major impact on the system performance and energy requirements. Note that when the direct heating contribution falls off during poor weather the majority of the difference is made up by auxiliary heat. Several comparisons can be made from Table 2 besides the deviations shown. For instance, assuming an electric rate of $.035/kwhr for State College and the system configuration given in Table 2, it would cost between $28 and $73 (depending upon the weather) to heat the house for the month of December.

TABLE 2 - A one month comparison of system performance characteristics for a solar assisted heat pump system in State College, Pa. ; Collector area = 1000 sqft and water storage capacity = 1000 gal

	Dec 1960 Weather Data	Dec 1972 Weather Data	Percent Deviation From 1960
Collector Efficiency (%)	36	44	22.2
Sys Coefficient of Performance	5.14	1.90	-63.0
Total Energy Collected (kwhr)	3648	1778	-51.2
Total Heating Load (kwhr)	4161	3950	-5.1
Heat Pump Contribution (kwhr)	2151	2508	16.6
Direct Heat Contribution(kwhr)	1902	193	-89.9
Aux Heat Contribution (kwhr)	100	1242	1142
Energy Input to Heat Pump (kwhr)	709	841	18.6
Energy Input to Aux Heat (kwhr)	100	1242	1142

Since the collector array is mounted on the roof of the residence, in reality an upper limit of collector array size is set by the area of the roof. Taking this into consideration a collector area of 600 sqft was used for the parametric studies of room thermostat setting and ethylene glycol/water heat exchanger size. Figure 14 shows for a collector area of 600 sqft the effect on SCOP of lowering the room temperature to 65 degrees F as the storage capacity is varied. It is interesting to note that the SCOP is increased when the room temperature is maintained at 65 degrees F versus 70 degrees F. Recalling that the SCOP is defined by equation (7) as

$$SCOP = \frac{\text{total heating load}}{\text{electric energy used by heat pump \& aux heater}} \qquad (7)$$

one might expect that since the heating load is reduced by about 13% the SCOP would decrease, but the reduction in energy input to the system is even greater. The heat pump input energy is reduced by 13% and the auxiliary heat input is reduced to nearly zero resulting in a net increase in the SCOP. As shown by the nearly horizontal curves of Figure 14, and as previously stated for a collector area of 600 sqft, the storage capacity size has little effect on the SCOP.

Table 3 shows how lowering the room temperature to 65

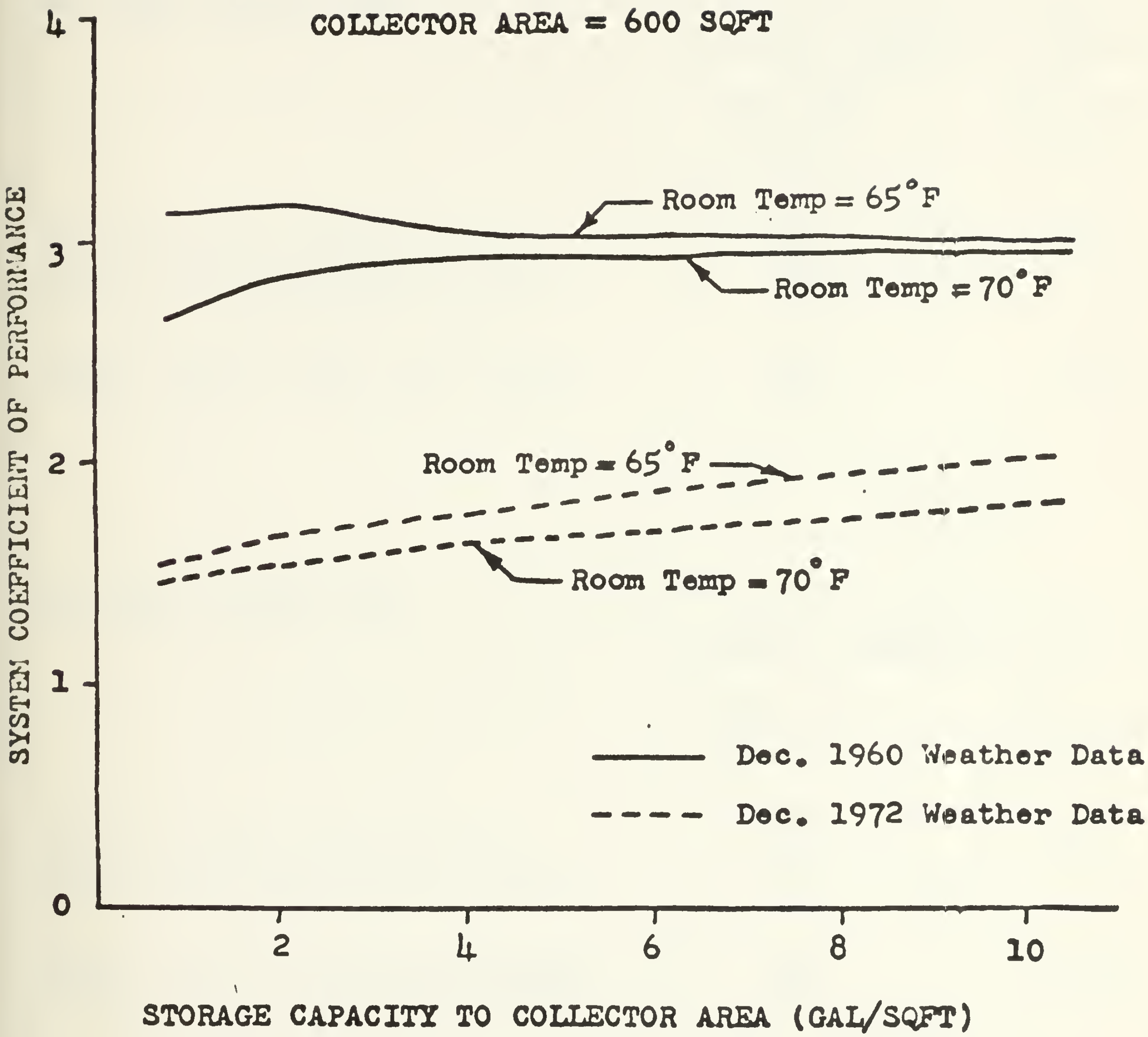

FIGURE 14 - Effect of room temperature on system coefficient of performance for a solar assisted heat pump system in State College, Pa.

TABLE 3 - Comparison of energy requirements for room temperatures of 65 F and 70 F for a solar assisted heat pump system in State College, Pa.

DEC 1960 WEATHER DATA

Collector Area = 600 sqft
Storage Capacity = 6000 gal

	Room Temp 65 F	Room Temp 70 F
Total Heating Load (kwhr)	3604	4161
Heat Pump Contribution (kwhr)	3598	4120
Direct Heat Contribution (kwhr)	0	0
Aux Heat Contribution (kwhr)	.03	34
Energy Input to Heat Pump (kwhr)	1193	1374
Energy Input to Aux Heat (kwhr)	.03	34

DEC 1972 WEATHER DATA

Collector Area = 600 sqft
Storage Capacity = 6000 gal

	Room Temp 65 F	Room Temp 70 F
Total Heating Load (kwhr)	3393	3950
Heat Pump Contribution (kwhr)	2622	2683
Direct Heat Contribution (kwhr)	0	0
Aux Heat Contribution (kwhr)	765	1263
Energy Input to Heat Pump (kwhr)	888	909
Energy Input to Aux Heat (kwhr)	765	1263

degrees F affects the energy inputs of the system. A storage capacity of 6000 gallons was chosen since Figure 10 shows that for 600 sqft of collector area the maximum SCOP is obtained with a storage capacity of 10 gal/sqft of collector area. The economics of lowering the room thermostat setting can be obtained from the values in Table 3. Again assuming an electric rate of \$.035/kwhr for State College it is seen that for the system configuration given in Table 3 approximately \$8 to \$18 can be saved (depending upon the weather) for the month of December by lowering the thermostat setting 5 degrees.

Tables 4 and 5 are the system performance characteristics for good (1960) and poor (1972) weather resulting from variations in the ethylene glycol/water heat exchanger effectiveness. If the capacity rate ratio is held constant at a value of 1.0 (i.e. $C_{min}/C_{max} = 1.0$) the heat exchanger effectiveness (E) is defined by the following equation:

$$E = NTU/(1+NTU) \qquad (8)$$

where NTU = Number of transfer units
(size parameter)

Thus by holding the capacity rate ratio constant at a value of 1.0, variations in the heat exchanger effectiveness (E) actually represent variations in the size of the heat

TABLE 4 - System performance factors of solar assisted heat pump system for varying values of heat exchanger effectiveness based on DEC 1960 weather data for State College, Pa.

HXGR EFF	STORAGE TANK TEMP(F)	COLLECTOR EFFICIENCY (%)	SYSTEM COP	CONTRIBUTION TO HEAT LOAD (%) HEAT PUMP	DIRECT HEAT	AUX HEAT
(Collector Area=600sqft Storage Capacity=600gal)						
.6	115.8	44.3	2.68	73	14	13
.7	117.3	44.7	2.71	72	15	13
.8	118.3	45.0	2.74	71	16	13
(Collector Area=600sqft Storage Capacity=2400gal)						
.6	87.1	47.7	2.93	99	0	1
.7	88.8	48.0	2.94	99	0	1
.8	90.0	48.3	2.95	99	0	1
(Collector Area=600sqft Storage Capacity=6000gal)						
.6	76.7	48.5	2.95	99	0	1
.7	78.0	49.0	2.96	99	0	1
.8	78.9	49.4	2.96	99	0	1

TABLE 5 - System performance factors of solar assisted heat pump system for varying values of heat exchanger effectiveness based on DEC 1972 weather data for State College, Pa.

HXGR EFF	STORAGE TANK TEMP(F)	COLLECTOR EFFICIENCY (%)	SYSTEM COP	CONTRIBUTION TO HEAT LOAD (%) HEAT PUMP	DIRECT HEAT	AUX HEAT
(Collector Area=600sqft Storage Capacity=600gal)						
.6	40.2	48.8	1.47	46	1	53
.7	39.8	49.4	1.48	46	1	53
.8	40.1	49.8	1.48	47	1	52
(Collector Area=600sqft Storage Capacity=2400gal)						
.6	40.0	52.6	1.62	57	0	43
.7	40.0	53.3	1.63	58	0	42
.8	40.0	53.8	1.63	58	0	42
(Collector Area=600sqft Storage Capacity=6000gal)						
.6	40.0	52.0	1.81	67	0	33
.7	40.0	52.7	1.82	68	0	32
.8	40.0	53.3	1.83	68	0	32

exchanger. The smaller the value of E the smaller is the size of the heat exchanger. As can be seen from Tables 4 and 5 varying the heat exchanger effectiveness (i.e. size) of the ethylene glycol/water heat exchanger has negligible effect on the performance of the system. The SCOP increases very slightly as the heat exchanger size increases regardless of the weather for the effectiveness range studied.

CHAPTER 4

4.1 Conclusions

The maximum collector efficiencies are obtained for system configurations with small collector areas and large storage capacities because the storage tank temperature and thus the temperature of the fluid entering the collector is kept low. Similarly, the collector efficiency of a given system is improved during periods of poor solar insolation due to the reduced entering fluid temperature. The collector efficiency increases as the storage capacity increases and as the collector area decreases since the combination of these result in a lower temperature of the fluid entering the collector. Weather extremes have little affect on the collector efficiency of systems with small collector area, however, the affect on the collector efficiency is significant for systems with large collector area. The study results also show that for the solar assisted heat pump system a storage capacity beyond 2 gal/sqft of collector area is of little benefit since the collector efficiency remains relatively constant.

For a solar assisted heat pump system the SCOP is limited to the heating COP of the heat pump unless the combination of good solar insolation and adequate collector

area result in a heating contribution from direct heating. The SCOP will be largest for systems with large collector areas and small storage capacities. There is a unique collector area at which the SCOP of the solar assisted heat pump system will equal the heating COP of the heat pump (i.e. 100% of the total residence heating requirement is provided by the heat pump) for a wide range of storage capacities. This unique collector area is dependent upon the solar insolation received and will vary in value according to the weather. For the solar assisted heat pump system modeled for State College in this study it is estimated that about 1200 sqft of collector area would be required to provide 100% heat pump heating under average December weather conditions.

The SCOP of the solar assisted heat pump system increases with increasing collector area and in general is more strongly influenced by the collector area than by the storage capacity. For a system with a collector area greater than that resulting in 100% heat pump heating, the residence heating load is supplied by contributions from the heat pump, auxiliary heat and direct heat and the SCOP decreases as the storage capacity increases. For a system with a collector area smaller than that resulting in 100% heat pump heating, the residence heating load is supplied essentially by the heat pump and auxilary heater and the

SCOP increases as the storage capacity increases. This situation suggests the use of a two tank storage system with a high temperature tank being used to maximize the direct heat contribution and a medium temperature tank for use as the heat pump heat source.

For a given collector area the percentage contribution of the heat pump increases as the storage capacity increases whereas the percentage contribution of the auxiliary heater decreases. Direct heating becomes a significant contribution to the total heating load for solar assisted heat pump systems with large collector areas and small storage capacities since the storage is heated to a high temperature level. Decreasing the collector area causes the percentage contribution of direct heating to decrease rapidly so that for small collector areas the direct heating contribution is non-existent.

When designing or simulating solar assisted heat pump systems it is essential that accurate and realistic weather data be used. The weather data used has a significant effect on the performance characteristics of the system. For the system modeled in this study the SCOP varied by as much as 250% for some collector/storage configurations when "poor" versus "good" weather data were used. The percentage contributions of the heat pump, auxiliary heater and direct

heating to the total heating load are similarly affected as was shown in Figure 13. A simple economic analysis of a system in State College with 1000 sqft of collector and 1000 gallons of storage shows that monthly heating costs can vary from $28 to $73 depending upon the weather.

Federal energy conservation guidelines call for reducing thermostat settings to 65 degrees F for winter heating. This study found that reducing the room temperature from 70 degrees F to 65 degrees F increased the SCOP approximately 5% and reduced significantly the amount of auxiliary heat required. For a system configuration of 600 sqft of collector area and 6000 gallons of storage capacity a simple cost analysis shows that in State College approximately $8 to $18 per month (depending upon the weather) can be saved by lowering the thermostat setting 5 degrees.

The performance of the solar assisted heat pump system is not significantly affected by the effectiveness (i.e. size) of the ethylene glycol/ water heat exchanger. Since this is the case, it appears that the heat exchanger could be of relatively simple design and low cost.

When reviewing the results presented in this study it must be remembered that several assumptions have been made

in modeling and simulating the operation of the residential solar assisted heat pump system. The majority of these assumptions are inherent in the TRNSYS components but others are externally controlled. For instance the effect of wind velocity has been assumed negligible which in a "real world" installation could reduce the solar energy collected by the collector. Also a ground reflectance value for bare ground has been assumed. In a "real world" installation the reflectance value might be higher if the ground was snow covered resulting in an increase of reflected solar radiation received by the collector. Overall, however, it is expected that the model performance will be somewhat more optimistic than that of a "real world" installation.

Although domestic hot water heating was not included in the solar assisted heat pump model of the residential heating system, the collector array sized for the solar assisted heat pump system would most likely provide all of the domestic hot water requirements in the summer months. During the heating season the system would be able to provide preheating of the domestic hot water much of the time.

At first glance the TRNSYS modeling program seems very complicated but actually the building block modeling

technique is fairly straightforward. Individual system components (i.e. pumps, solar collectors, storage tank, etc.) are described by component models which are connected together to form a complete system. It is better to keep the model as simple as possible since problems, such as failure of the differential equation solution techniques to converge within the component models, are likely to result if the model is too elaborate. Some modes of the TRNSYS components use empirical equations which have been developed in SI units requiring the entire simulation to be run in SI units if these component modes are used. If these component modes can be avoided in the model the more familiar English units can be used. The TRNSYS simulations require hourly weather data as input. The assembly and handling of such data can become quite cumbersome if punched cards are being used, especially if the simulation is run for a full year. This problem can be alleviated by using a tape input. The computer time required for running a full year simulation of a solar assisted heat pump system (based on this one month run) will be about 200 seconds and cost approximately $60 if output is printed for each hour.

4.2 Recommendations

To obtain a complete picture of the solar assisted heat pump system, a detailed economic analysis of the system

alternatives should be performed. This would determine the most cost effective combination of such things as collector area and storage capacity. An economic analysis would also be an effective method to compare the solar assisted heat pump system with conventional heating systems.

An entire year or at least a complete heating season of weather data should be used when simulating the solar assisted heat pump system rather than a single month. This will give a more realistic picture of the system's seasonal performance characteristics. Limiting the weather data to the heating season will significantly decrease the amount of data and reduce computer costs. It is important however that the weather data be carefully selected to provide a realistic representation of the location being studied.

Future studies on the solar assisted heat pump system should explore the feasibility and operating characteristics of a two tank storage system for optimizing the output of the solar assisted heat pump system. Also the results presented in this report and any future simulations using the TRNSYS program should be compared with other modeling techniques such as those being developed by the Architectural Engineering Department of the Pennsylvania State University.

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APPENDIX A

BUILDING CRITERIA

Indoor Design Temperature = 70F

Outdoor Design Temperature = 7°F

CONSTRUCTION DATA

Wall:
4" Face Brick
3/4" Air Gap (Mean Temp. = 50, Temp. Diff. = 10, E = .82)
Building Membrane (Permeable Felt)
1/2" Plywood (Douglas Fir)
3 1/2" Fiberglass Insulation
1/2" Gypsum Wall Board
2 x 4 Stud Wall Construction, 16" O.C.

Glass:
Area is given on plan; e.g. A = 15
All Double Pane
(1/4" Air Gap, Wood Sash, 80% Glass)

Doors:
3' x 7'
1 1/2" Solid Wood
Metal Storm Door (Front Door only)

Ceiling:
6" Fiberglass Insulation
1/2" Gypsum Board
2 x 6 Ceiling Joists, 16" O.C.

U-Values:

Walls:	R	1977 Handbook of Fundamentals Reference Page No.	Table No.
Outside Air Film (15 mph wind)	.17	22.11	1-A
4" Face Brick	.44	22.15	3-A
3/4" Air Gap (Mean Temp. = 50, Temp. Diff.= 10, E = .82)	1.01	22.12	2
Building Membrane (Permeable Felt)	.06	22.14	3-A
1/2" Plywood (Douglas Fir)	.62	22.13	3-A
3 1/2" Fiberglass Insulation	11.00	22.14	3-A
1/2" Gypsum Wall Board	.45	22.13	3-A
Inside Air Film	.68	22.11	1-A
TOTAL RESISTANCE	14.43		
FINAL U-VALUE	.069		

Glass:	U		
Double Pane (1/4" Air Gap)	.58	22.24	8-A
Adjustment for Wood Sash, 80% Glass (Ux adjustment)	.95	22.24	8-C
FINAL U-VALUE	.55		

Door:	U		
1 1/2" Solid Wood .49 with metal storm door	.33	22.25	9
FINAL U-VALUE	.33		

Ceiling:	R		
Air Film (Still Air, E = .90)	.61	22.11	1-A
6" Fiberglass Insulation	22.00	22.14	3-A
1/2" Gypsum Wall Board	.45	22.13	3-A
Inside Air Film	.61	22.11	1-A
TOTAL RESISTANCE	23.67		
FINAL U-VALUE	.042		

Kitchen Wall Against Garage:	R		
Air Film (Still Air, E = .90)	.68	22.11	1-A
3 1/2" Fiberglass Insulation	11.00	22.14	3-A
1/2" Gypsum Wall Board	.45	22.13	3-A
Inside Air Film	.68	22.11	1-A
TOTAL RESISTANCE	12.81		
FINAL U-VALUE	.078		

HEAT LOSS SUMMARY

Location (City) STATE COLLEGE, PA
Design IDBT = 70F
Design ODBT = 7° F　　Design Temp. Diff. = 63 ° F
Type of Building 2 STORY RESIDENTIAL

STRUCTURAL COMPONENT	AREA (SQ FT)	U-VALUE	UA	HEAT LOSS, Btuh	
1ST FLOOR					
WALLS	942	.069	65.00	4095	
GLASS	141	.55	77.55	4886	
FLOOR	ASSUME NO HEAT LOSS				
DOORS	21 21	.33 (w/STORM DOOR) .49	6.93 10.29	1085	
INTERIOR WALL	108	.078	8.42	530	ASSUME ΔT=63°
SECOND FLOOR					
WALLS	1122	.069	77.42	4877	
GLASS	111	.55	61.05	3846	
CEILING	1105	.042	46.41	2924	ASSUME ΔT=63°
			SUB TOTAL	22243	

17307 FT³
INFILTRATION: No. of Air Changes 144 CFM 1/2/HR　　9979
$Q = 1.1(CFM)\Delta T$

POSITIVE VENTILATION: 0 CFM　　—

TOTAL　　32222

$$\frac{\text{HEAT LOSS}}{70 - \text{ODBT}} = \frac{32222}{70-7} \quad \frac{\text{Btuh}}{{}^\circ\text{F ind-out temp diff}} = 511$$

COOLING LOAD CALCULATION

SOURCES: Chap. 22, 1972 ASHRAE Handbook of Fundamentals
Chap. 25, 1977 ASHRAE Handbook of Fundamentals

Location (City) STATE COLLEGE, PA

Design Conditions 75 IDBT, 87 ODBT

Daily Range 23 F

Design Equivalent Temperature Differences:

Brick Walls 8.3 F — Ceiling & Roofs 36 F

Wood Doors 15.6 F — Floors 0 F

Design Window Factors:

North 13 Btuh/ft² — West 43 Btuh/ft²

East 43 Btuh/ft² — South 21 Btuh/ft²

COOLING LOAD SUMMARY

Walls	1182	Btuh
Windows	5916	"
Doors	321	"
Ceiling	1671	"
Floor	0	"
Occupants	900	"
Cooking	1200	"
ALL AREA) (.9 BTUH/FT²) Infiltration	2114	"
Total	13304	Btuh, Total Sensible Heat (TS

Total Heat = 1.3 (TSH) = 17295 Btuh

Equipment Size = $\frac{\text{Total Heat, Btuh}}{\text{12,000 Btuh/ton}}$ = 1.44 tons

APPENDIX B

DEFINITION OF TERMS USED IN TRNSYS COMPUTER MODEL FLOW DIAGRAM

DELTAU - Total change in storage tank internal energy

H - Solar radiation (total) on a horizontal surface

Hb - Solar radiation (beam) on a horizontal surface

Hd - Solar radiation (diffuse) on a horizontal surface

HT - Solar radiation (total) on a tilted surface

HB - Solar radiation (beam) on a tilted surface

HD - Solar radiation (diffuse) on a tilted surface

H1 - Fraction of timestep that heat pump operated in water-air heating mode

H2 - Fraction of timestep that heat pump operated in air-air heating mode

H3 - Fraction of timestep that heat pump operated in cooling mode

IGAM - Heat pump master control integer; 0=off, 1=on

Mi - Flowrate of water entering heat pump evaporator

Mo - Flowrate of water returning from heat pump evaporator

MODE - Mode of heat pump operation

OPHRS - Total time heat pump operated in heating mode

QA1 - Instantaneous energy absorbed from liquid source in water-air heating mode

QABS - Total energy absorbed by heat pump

QAC - Instantaneous energy removed from load in cooling mode

QAUXH - Auxiliary energy delivered to room in heating mode

QDH - Energy delivered to room in direct heating mode

QDUMP - Rate at which energy is discarded by relief valve

QENV - Rate of energy loss from storage tank to surroundings

QL - Instantaneous room heating load

QLOAD - Hourly heating load

QR1 - Instantaneous energy delivered to load in water-air heating mode

QR2 - Instantaneous energy delivered to load in air-air heating mode

QRC - Energy rejected to sink in cooling mode

QREJ - Total energy delivered by heat pump

QSHRTC - Cooling capacity required but not met by heat pump

QTANK - Rate of energy delivery from storage tank to heat pump

Qu - Rate at which energy is transferred to storage tank

QUSE - Hourly useful energy collected

QW - Total work input to heat pump

RAD - Hourly incident solar radiation on collector

T - Temperature of storage tank

Ta - Ambient temperature

TENV - Temperature of ground surrounding storage tank

Ti - Temperature of water entering heat pump evaporator

To - Temperature of water leaving heat pump evaporator

TOTQL - Total heating load

TOTQU - Total useful energy collected

TOTRAD - Total incident solar radiation on collector

TSTOR - Hourly storage tank temperature

WAC - Instantaneous work input to heat pump in cooling mode

WAH - Instantaneous work input to heat pump in heating mode

WORKIN - Total work input to heat pump in heating mode

ΔE - Change in internal energy of water in storage tank

θt - Angle of incidence of solar radiation on a tilted surface

TYPICAL TRNSYS COMPONENT PARAMETERS

(A) UNIT 1, TYPE 9 READER

Parameters		Description	Value
1	N	number of values to be read	2
2	Δt	time interval at which data is provided	1 hr
3	i	position of value on card for which units are to be converted	2
4	Mi	multiplication factor for that value	3.686
5	Ai	addition factor for that value	0

(parameters 3,4&5 convert Langleys to Btu/hr-sqft)

6	L	logical unit number of data file	8

(B) UNIT 2, TYPE 16 RADIATION PROCESSOR, MODE 3

Parameter		Description	Value
1		mode	3
2	n	day of year at start of simulation	335
3	ϕ	latitude	40.5 deg
4	s	slope of collector surface	50.5 deg
5	γ	orientation angle of collector	0 deg
6	Sc	solar constant	429 Btu/hr-sqft
7	ρ	ground reflectance	.2

(C) UNIT 3, TYPE 21 LIQUID COLLECTOR-STORAGE MODULE

Parameter		Description	Value
1	A	collector area	600 sqft
2	$\dot{M}c$	collector fluid flowrate	7500 lb/hr
3	Cpc	collector fluid specific heat	.792 Btu/lb-F
4	$\dot{M}h$	heat exchanger to tank flowrate	5940 lb/hr
5	Cph	specific heat of fluid in storage tank	1.0 Btu/lb-F
6	F'	collector efficiency factor	.99
7	$\tau\alpha$	transmittance-absorptance product	.684
8	UL	collector loss coefficient	.7 Btu/hr-sqft-F
9	Tmax	temperature at which tank relief valve opens	212 deg F
10	ϵ	collector-tank heat exchanger effectiveness	.7
11	V	volume of storage tank	80.1 cuft
12	ρ	tank fluid density	62.4 lb/cuft
13	U	tank energy loss coefficient	.05 Btu/hr-sqft-F
14	r	ratio of tank height to tank diameter	3

15	To - initial tank fluid temperature	70 deg F

(D) UNIT 4, TYPE 20 WATER-TO-AIR HEAT PUMP

Parameter	Description	Value
1	CP1 - specific heat of liquid entering heat exchanger A	1.0 Btu/lb-F
2	MDOT1 - mass flow rate of liquid entering heat exchanger A	2500 lb/hr
3	TROOMH - constant room temperature for heating mode	70 deg F
4	TROOMC - constant room temperature for cooling mode	77 deg F
5	TMIN1 - minimum liquid source temp for water-to-air heat pump	40 deg F
6	TMIN2 - minimum air source temp for air-to-air heat pump	1000 deg F
7	NDATAH - number of equally spaced heating data points	8
8	NDATAC - number of equally spaced cooling data points	8
9	ISAVEH - logical unit number of heating data file	31
10	ISAVEC - logical unit number of cooling data file	32
11	TCOOL - minimum ambient temp when cooling is allowed	1000 deg F
12	CAIR - MCp product for room air flow thru heat exchangers B,C and D	648 Btu/hr-F
13	EFF - effectiveness of heat exchanger B	.7
14	TSET - minimum liquid source temp for direct heating from liquid source	110 deg F
15	ICOOL - cooling condenser selection integer, 0=heat exchanger A	0
16	THEAT - maximum ambient temp when heating is allowed	65 deg F

(E) UNIT 5, TYPE 12 HEATING LOAD, MODE 3

Parameter	Description	Value
1	mode	3
2	UA - heating requirement of house (energy/degree hour)	-511 Btu/hr-F
3	TR - room temperature	70 deg F

(F) UNIT 6, TYPE 24 QUANTITY INTEGRATOR

No parameters required

(G) UNIT 7, TYPE 25 PRINTER (HOURLY)

Parameter	Description	Value
1	Tp - time interval at which printing is to occur	1 hr

(H) UNIT 8, TYPE 25 PRINTER (24 hour)

Parameter	Description	Value
1	Tp - time interval at which printing is to occur	24 hr

APPENDIX C

```
              TRNSYS - A TRANSIENT SIMULATION PROGRAM
        FROM THE SOLAR ENERGY LAB AT THE UNIVERSITY OF WISCONSIN
                       VERSION 8.1  9/27/76

SIMULATION          0.0            7.440E 02    2.500E-01

TOLERANCES          -1.000E-02    -1.000E-02

UNIT  1     TYPE  9       WEATHER DATA READER
    PARAMETERS  6
    2.000E 00     1.000E 00     2.000E 00     3.686E 00     0.0             8.000E 00

UNIT  2     TYPE 16       RADIATION PROCESSOR, MODE 3
    PARAMETERS  7
    3.000E 00     3.350E 02     4.050E 01     5.050E 01     0.0             4.290E 02     2.000E-01
    INPUTS  1
     1, 2
    0.0

UNIT  3     TYPE 21       LIQUID COLLECTOR-STORAGE MODULE
    PARAMETERS 15
    6.000E 02     7.500E 03     7.920E-01     5.940E 03     1.000E 00     9.900E-01     6.840E-01     7.000E-01
    2.120E 02     7.000E-01     8.010E 01     6.240E 01     5.000E-02     3.000E 00     7.000E 01
    INPUTS  5
     2, 1          1, 1          4, 1          4, 2          0, 0
    0.0           2.800E 01     7.000E 01     2.500E 03     5.000E 01

UNIT  4     TYPE 20       WATER TO AIR HEAT PUMP
    PARAMETERS 16
    1.000E 00     2.500E 03     7.000E 01     7.700E 01     4.000E 01     1.000E 03     8.000E 00     8.000E 00
    3.100E 01     3.200E 01     1.000E 03     6.480E 02     7.000E-01     1.100E 02     0.0           6.500E 01
    INPUTS  5
     3, 1          3, 2          1, 1          5, 3          0, 0
    7.000E 01     2.500E 03     2.800E 01    -2.146E 04     1.000E 00

UNIT  5     TYPE 12       HEATING LOAD(ENERGY/DEGREE HOUR MODEL), MODE
    PARAMETERS  8
    3.000E 00    -5.110E 02     7.000E 01     0.0           0.0           0.0           0.0           0.0
    INPUTS  4
     0, 0          0, 0          1, 1          0, 0
    0.0           0.0           2.800E 01     0.0

UNIT  6     TYPE 24       QUANTITY INTEGRATOR
    INPUTS 12
     2, 1          3, 3          3, 4          3, 5          3, 7          4, 3          4, 4          4, 5
     4, 6          4, 9          4,10          5, 3
    0.0           0.0           0.0           0.0           0.0           0.0           0.0           0.0
    0.0           0.0           0.0          -2.146E 04

UNIT  7     TYPE 25       PRINTER(HOURLY)
    PARAMETERS  1
    1.000E 00
    INPUTS  4
     2, 1          3, 1          3, 3          5, 3
    RAD           TSTOR         QUSE          QLOAD

UNIT  8     TYPE 25       PRINTER(24 HOUR)
    PARAMETERS  1
    2.400E 01
    INPUTS 13
     3, 6          6, 1          6, 2          6, 3          6, 4          6, 5          6, 6          6, 7
     6, 8          6, 9          6,10          6,11          6,12
    DELU          TOTRAD        TOTUU         QENV          QTANK         QDUMP         WORKIN        QABS
    QREJ          HRS           QDH           QAUX          TOTQL
END

      EXECUTE TRNSYS                                                              00003300

   ***MESSAGE 5.11***    SUBSYSTEM TRNSYS - VERSION 8.1  05/01/78
```

```
TRANSIENT SIMULATION      STARTING AT TIME = 0.0
                          STOPPING AT TIME = 7.440E 02
                                  TIME STEP = 2.500E-01
      DIFFERENTIAL EQUATION ERROR TOLERANCE =-1.000E-02
          ALGEBRAIC CONVERGENCE TOLERANCE =-1.000E-02

                                       TIME =     0.0
   RAD        TSTOR      QUSE        QLOAD       DELU       TOTRAD     TOTQU      QENV       QTANK
0.0        7.000E 01   0.0        -2.146E 04   0.0        0.0        0.0        0.0        0.0
  QDUMP       WORKIN     QABS        QREJ        HRS        QDH        QAUX       TOTQL
0.0        0.0         0.0        0.0          0.0        0.0        0.0        -2.146E 04

                 TIME =    1.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        6.750E 01   0.0        -2.146E 04

                 TIME =    2.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        6.466E 01   0.0        -2.146E 04

                 TIME =    3.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        6.179E 01   0.0        -2.197E 04

                 TIME =    4.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        5.893E 01   0.0        -2.146E 04

                 TIME =    5.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        5.609E 01   0.0        -2.197E 04
                    •

                    •

                    •

                    •

                 TIME =  742.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        1.229E 02   0.0        -1.380E 04

                 TIME =  743.0000
   RAD        TSTOR      QUSE        QLOAD
0.0        1.201E 02   0.0        -1.329E 04

                                       TIME =  744.0000
   RAD        TSTOR      QUSE        QLOAD       DELU       TOTRAD     TOTQU      QENV       QTANK
0.0        1.173E 02   0.0        -1.380E 04   2.346E 05  3.451E 04  9.264E 06  1.338E 05  8.894E 06
  QDUMP       WORKIN     QABS        QREJ        HRS        QDH        QAUX       TOTQL
0.0        3.399E 06  -6.764E 06  1.021E 07   3.908E 02  2.133E 06  1.832E 06  -1.420E 07


              UNIT  1 WAS CALLED  2978 TIMES
                    2             2978
                    3             9010
                    4             6407
                    5             2978
                    6             2976
                    7              745
                    8               32

  ******************************
  *        END OF RUN          *
  ******************************

  TIME = 21:32:50.82

  ELAPSE TIME =   18.88 SEC.
```

Thesis
D78877
c.1

Dunbar

187979

Analysis of a residential heating system utilizing a solar assisted water-to-air heat pump.